EXHIBIT A

Program 60-1161—Simple Epicyclic Gear Design (Parallel Axis)

Introduction

The simple epicyclic gear unit consists of acentral external gear (sun gear) meshed with one or more external gears (planet gears). The planet gears are then meshed with an internal gear (ring gear) which encloses the system. The planet gears and planet gear support bearings are usually held in a carrier which rotates about the geometric center of the unit. The term "epicyclic" comes from the path of a point on a planet gear which traces out an epicycloid in space. There are three basic types of simple epicyclic gear units.

When the ring gear is fixed or grounded and the sun and carrier are input/output members the unit is called a "planetary gear".

When the carrier is fixed and the sun and ring are input/output members the unit is a "star" gear. The star gear is not an epicyclic gear as the planet gear centers do not rotate about the unit central axis but since the construction is basically the same it is included in the family. For high speed units the star gear is often used in cases where planetary gears are not practical because of the high centripetal acceleration loads on the planet gears.

When the sun is fixed and the carrier and ring are input/output members the unit is a "solar" gear.

The range of speed reduction ratios for which these units can be designed with reasonable proportions is as follows:

Planetary Gear:	3:1 to 12:1
Star Gear:	2:1 to 11:1
Solar Gear:	1.2:1 to 1.7:1

Below these ranges the planet gears become quite small and it becomes difficult to design the gears and planet bearings for reasonable life. Above these ranges the sun gear becomes small and the number of planets that can be used without interference is limited. This, again, makes the design of the bearings difficult. Ratios between 1.7:1 and 2:1 are difficult to design successfully with simple epicyclic gearing, although it can be done with compound epicyclic units. (See UTS model 60-1162 for Compound Epicyclic Gearing.)

Epicyclic units are often used as differentials. UTS model 60-1161 is restricted to systems in which one element is fixed and does not consider use as a differential. UTS models 60-1163 and 60-1164 treat simple and compound epicyclic units used as differentials.

If more than one planet gear is used the number of planets that will assemble between the sun and ring is limited by the numbers of teeth in the sun and ring and by the possibility of interference between the tips of the planet gear teeth. For a number of planets to assemble equally spaced around the center, the sum of the tooth numbers in the ring and sun divided by the number of planets used must be an integer:

$$(N_{ring}+N_{sun})/n_p = \text{integer}$$

where: N_{ring} = Number of teeth in ring gear
N_{sun} = Number of teeth in sun gear
n_p = Number of planet gears

The distance between the planet gear centers in the carrier must, of course, be greater than the outside diameter of the planet gears or tooth tip interference will result (assuming the planet gears are in the same plane).

It is not necessary that the planets be equally spaced. However, to make assembly possible, they must be spaced at multiples of the "Least mesh angle".

$$e_p/\beta = \text{integer}$$

$$\beta = 360°/(N_{ring}+N_{sun})$$

where: e_p = Angle between adjacent planet gears, deg
β = Least mesh angle, deg

For example, suppose we have an epicyclic set with N_{ring} = 68 teeth and N_{sun} = 18 teeth and we wish to use 4 planets arranged 90 degrees apart. $(N_{ring}+N_{sun})/4 = 21.5$, which is not an integer so we cannot arrange 4 planets 90 degrees apart. $(N_{ring}+N_{sun})/2 = 43$ which is an integer so we can arrange 2 planets 180 degrees apart. The least mesh angle, β =360 degrees/$(N_{ring}+N_{sun})$ = 4.186 degrees. When we attempt to place a planet 90 degrees from the first planet we find that we are at 90 degrees/ β = 21.5 least mesh angles and cannot assemble. We can, however, place the planet at 21 or 22 least mesh angles. This would put the planet gear at .5 β or 2.093 degrees from 90 degrees. Then, because we know that 2 planets will assemble 180 degrees apart, the 4 planets would be placed at 0 degrees, 87.907 degrees, 180 degrees and 267.907 degrees. The tip clearance should then be checked. Because we have two sets of planets 180 degrees apart the (theoretical) summation of the bearing loads on the sun and ring is still zero.

It is not necessary (or even desirable) that $N_{ring} = N_{sun} + 2 \cdot N_{planet}$. If this relationship is met and the center distance is "standard" then the operating pressure angles at the sun/planet external mesh, ϕ_{ext}, and the planet/ring internal mesh, ϕ_{int}, will be equal to the nominal pressure angle of the system. If ϕ_{ext} is made higher than nominal and ϕ_{int} lower than nominal it will increase the strength of the set and reduce the burst stress on the ring. ϕ_{ext} and ϕ_{int} can be easily controlled by the number of planet teeth and the operating center distance.

Examples

If you are using model 60-1161 for the first time you may wish to run the following example.

Assume we wish to design a spur gear planetary set with about 4.95 to 1 reduction ratio with a ring gear diameter of about 8 inches. Also, assume the smallest number of teeth we wish to use is about 17. (The number of teeth would be selected according to material and duty cycle;-see UTS model 60-180.)

In the wizard data input form, enter 17 in the input column for "Sun Gear Teeth" and 4.95 for "Planetary Gear" ratio. The data input form is shown in Figure 1, the solution in Report 1.